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SUMMARY

A small-scale experiment is reported which demonstrated that a few closely spaced thin-film plastic "bags," hydraulically connected and mounted close to the propellant tank wall with liquid hydrogen in all volumes, could act as sufficient insulation to cause liquid in the volume nearest the tank wall to boil off sacrificially much of the incoming heat leak. Thus, much of this heat leak is prevented from reaching the liquid shielded by this insulator. As a consequence, the temperature of the liquid in a pressurized tank increases at a much slower rate than it would if there were no bags. In this case, the heat leak to the shielded liquid is essentially independent of the incoming heat leak to the tank and depends only on the tank pressure. The lowest heating rates are possible for low tank pressures. This method could result in a weight saving for booster rocket vehicles designed for its application because it can reduce the pump cavitation problem by decreasing the liquid heating rate. Further weight saving appears to be possible in pressurization system weight. The greatest overall weight saving occurs at low tank pressures.

INTRODUCTION

In pump-fed rocket systems that use cryogenic liquids as propellants, the liquid must be supplied to the inlet of the main pump with a net positive suction head greater than some minimum critical value in order to prevent deterioration in propellant flow by pump cavitation (the net positive suction head is the excess of tank pressure over liquid vapor pressure). A large part of the required net positive suction head is obtained by subcooling the liquid; that is, the liquid temperature is lower than the boiling temperature. Subcooling is obtained by pressurizing the liquid to a pressure greater than the vapor pressure of the liquid. In rocket systems that use cryogenic propellants, subcooling is achieved either by pressurizing the propellant tank or by using a boost pump ahead of the main pump.

Generally, pressurization of tanks for structural reasons is required for a considerable period of time before the start of propellant consumption. During this time the heat leak from the environment raises the liquid temperature and thus decreases the available net positive suction head, which may decrease sufficiently to cause cavitation during propellant consumption. This

liquid heating problem is accentuated because the liquid temperature is not raised uniformly. Most of the heat entering the tank is carried by convection currents into a warm liquid layer below the liquid-vapor interface thereby producing temperature stratification (refs. 1 and 2). The temperature of this layer approaches that of saturation, and therefore it possesses a low net positive suction head. The layer grows with time and is not destroyed during propellant outflow. In booster vehicles, where the heat leak can be high, this low net positive suction head layer can become quite deep. Consequently, a large amount of propellant could be unusable near the end of propellant consumption by the loss of propellant flow due to detrimental pump cavitation (ref. 3).

Several approaches have been proposed or used to cope with the problems of propellant temperature stratification. The tank wall insulation can be increased to reduce the heat leak. The tank operating pressure can be increased to give a higher degree of liquid subcooling initially. The propellant tank size can be increased to carry more propellant and thereby compensate for that lost as a result of temperature stratification. The use of a boost pump, which can handle a boiling liquid ahead of the main pump, side-steps the liquid heating problem. All these methods can add considerable weight to the rocket system.

This report presents and evaluates another method for decreasing the propellant heating rate. An internal "tank" made of insulating material (a thermal and convection barrier) is placed inside the tank so that a small volume of liquid exists between the barrier and the tank wall. Heat transferred into the tank vaporizes the liquid in this volume next to the wall. The vapor thus generated is vented to maintain a prescribed tank pressure and carries with it much of the heat that enters the tank. The amount of heat that reaches the bulk liquid depends on the tank pressure and on the thermal conductance of the barrier. Although venting results in loss of propellant, this method could result in a lower weight penalty than the other methods mentioned previously because, for the same total heat leak, the mass of liquid vaporized and vented is small compared with the mass of liquid that could be unusable because it was heated sufficiently to cause detrimental pump cavitation.

Since conventional insulating materials for the barrier may not be practical from structural and weight standpoints, a preliminary analytical and experimental investigation of the use of the propellant itself as the insulating material was conducted at the Lewis Research Center. The thermal barrier investigated consisted of three thin plastic "bags," one inside the other, with propellant contained between them. The spacing between the bags was chosen to reduce convection currents within the bags. The weight of this scheme is essentially that of the bags because the propellant in the outer volume and within the bags can be consumed as the liquid there mixes uniformly with the liquid enclosed by the bags during outflow.

The objective of the investigation was to establish the feasibility of using sacrificial boiloff and a thermal barrier, composed of nearly saturated propellant, in order to decrease the rate of heating of the liquid propellant. No effort was made to develop a practical thermal barrier for flight use, although a technique for this application is suggested. It is also pointed

out that pressurization gas requirements might be reduced by using such a barrier.

PROPELLANT TEMPERATURE STRATIFICATION PROBLEM

The processes that occur in a cryogenic tank to produce propellant liquid temperature stratification are illustrated schematically in figure 1. After pressurization of the filled propellant tank, for structural purposes and/or to subcool the cryogenic liquid, heat flux to the liquid goes mainly to raising the liquid temperature. Much of the heat leak through the tank side walls is transported by convection currents up the walls of the tank toward the liquid-vapor interface, where it forms a growing layer of relatively warm liquid, whereas the heat leak through the tank bottom and part of the side wall heat leak tend to heat the propellant more uniformly (ref. 2), as indicated by the temperature profiles in figure 1. Interfacial heat and mass transfer also add to the warm layer. The liquid-vapor interface temperature corresponds very closely to that of saturation for the tank pressure. Since the density of the warm layer is less than that of the bulk liquid, the temperature stratification is stable, and the layer is not destroyed during propellant outflow (refs. 1 and 2). When the warm, nearly saturated, liquid layer arrives at the pump inlet, detrimental pump cavitation may occur, resulting in reduced flow or flow stoppage. In booster vehicles, where the heat leak is high, the warm layer can become quite thick, so that a large amount of propellant might not be usable.

The present methods used to reduce the liquid temperature stratification are to increase the amount of insulation and/or increase the tank operating pressure. Increasing the amount of insulation reduces the heat leak, while increasing the tank pressure increases the amount of heat that can be absorbed by the liquid before saturation conditions are reached (i.e., increased initial net positive suction head). The effectiveness of these methods and their limitations are best shown by the use of a highly idealized illustrative example. Consider an insulated cylindrical tank (length, 30 ft; diam 15 ft; 90 percent full of liquid hydrogen) 2 minutes after it has been rapidly pressurized from 15 pounds per square inch absolute to a working pressure p (all symbols are defined in appendix A). Assume that all the heat leak to the liquid during the 2 minutes goes into a warm layer below the liquid-vapor interface, which is assumed to be saturated. Assume further that this warm layer of liquid is unusable due to detrimental pump cavitation. The ratio of the usable propellant mass to the filled tank mass η at the end of this time period is calculated for some idealized tank configurations in appendix B. The usable propellant mass fraction η is plotted as a function of tank pressure in figure 2 for the tank configuration and parameters indicated in this figure and in appendix B. Cases A compare the effect of tank pressure on η for a number of insulation thicknesses ($d = 1/8$, $1/4$, 1, and 2 in.). These curves indicate that the usable propellant mass fraction η rapidly increases as the tank pressure increases up to a maximum beyond which the mass fraction decreases because the tank walls are thicker. Increasing the insulation thickness shifts the peak mass fraction η toward a lower pressure. The curves also indicate that the mass fraction reaches a maximum at an insulation thickness between 1 and 2 inches.

It would seem desirable, from a weight standpoint, to operate at a higher value of η. This could conceivably be accomplished if there were some means of completely converting incoming heat leak into boiloff vapor without heating the bulk liquid. From a weight standpoint, it might be more efficient to dispose of heat leak by sacrificially boiling off some liquid rather than allowing the liquid to heat, with the result that liquid would be lost by detrimental pump cavitation. For example, consider a tank that is rapidly pressurized to a pressure p from an initial equilibrium condition at a pressure p_o of 15 pounds per square inch absolute, while somehow the bulk liquid remains at its initial temperature T_o. The usable propellant fraction η for this ideal situation is plotted in figure 2 as case B.

Apparently sacrificial boiloff could result in a considerable propellant weight saving at low tank pressures provided that the bulk liquid is not heated. A proposed method to take advantage of sacrificial boiloff is described subsequently.

Thermal Barrier

Basic concept. - Consider a tank whose liquid is initially at T_o and p_o that is suddenly pressurized to a pressure p prior to outflow. The liquid in a propellant tank can be largely shielded from heat leak by placing insulation inside the tank, a small distance from the tank wall (fig. 3) so that the liquid enclosed by the insulation does not mix with the liquid outside. The insulation acts as a sufficient thermal resistance to cause the liquid in the outer volume to boil off much of the heat input to the tank so that the temperature $T_l(t)$ of the liquid shielded by this insulation rises more slowly than if there were no such insulation. The temperature differences across the insulation ΔT_b, where the thermal storage of the insulation is negligible, dictates the heat leak to the shielded liquid. Incoming heat leak quickly raises the temperature of the small outer liquid volume to saturation $T_{s(p)}$. Thereafter, heat leak to the liquid shielded by the insulation will be independent of the incoming heat leak and dependent only on ΔT_b, which in turn depends only on the tank pressure p and the temperature of the bulk liquid (i.e., $\Delta T_b = T_{s(p)} - T(t)$). For low tank pressures, the maximum temperature difference $\Delta T_{b,max} = T_{s(p)} - T_o$ can be small so that heat leak to the shielded liquid could conceivably be much smaller than the incoming heat leak. The effectiveness of this method to reduce liquid heating is indicated in the following paragraph.

Instead of adding an 1/8 inch of insulation to the outside of the tank of case A to form a 1/4-inch-thick insulation, the 1/8 inch of insulation is placed inside the tank, as in figure 3, to form a thermal barrier. This situation is analyzed in appendix B and is plotted as case C in figure 2. Apparently this inner insulating liner gives very nearly the ideal performance of case B, and consequently an appreciable weight saving appears possible.

This concept is limited by the weight and practicality of the insulation used inside the tank. For reasons of practicality and weight, a rigid insulator of solid or evacuated insulation may not be a good choice. Some type of

insulator differing from these should be considered.

Liquid thermal barrier. - When a fluid is confined in a narrow space, it tends to become stagnant, and its thermal conductance approaches that predicted by the thermal conductivity of the fluid itself, since convection heat transfer is decreased as the spacing narrows (ref. 4). Both liquid and gaseous hydrogen have a low thermal conductivity (K_l = 0.07, K_g = 0.01 (Btu)(ft)/(sq ft)($^{\circ}$R)(hr). A proposed arrangement to take advantage of this property is shown in figure 4. The insulator shown (thermal barrier) is composed of three closely spaced thin-film plastic bags, which are mounted near the wall of the tank. The bags are hydraulically connected at their top and bottom to equalize the liquid level and to minimize the loads on them caused by pressure differences, which would occur during tank pressurization, filling, sloshing, outflow, and flight. The hydraulic connections are designed to minimize the mixing of outer volume liquid with the barrier and the shielded liquid.

In the sections that follow, the overall thermal conductance of liquid hydrogen, confined between layers of thin-film plastic, will be evaluated analytically and experimentally. The reduction in the problem of liquid heating and cavitation, made possible by the thermal barrier, will be estimated.

Thermal conductance: The heat leak through the thermal barrier can be predicted by

$$\left(\frac{Q}{A}\right)_b = U_b(\Delta T_b) \tag{1}$$

where the overall thermal conductance of the barrier U_b is defined by equation (1). For one-dimensional heat flow, where the effect of thermal storage in the barrier layers is assumed to be negligible, the thermal conductance U_b is given by

$$U_b = \frac{1}{\dfrac{1}{\mathcal{H}_{sl}} + \displaystyle\sum_{i=1}^{N} \frac{\delta_i}{K_{eff,i}} + \dfrac{1}{\mathcal{H}_{s'}}} \tag{2}$$

The term $K_{eff,i}$ takes into account the heat transfer by conduction, convection, and radiation across the barrier. The surface coefficients of heat transfer $\mathcal{H}$ at the outermost bag walls of the barrier can, in this case, be safely assumed to be large (e.g., $\mathcal{H}$ > 100 Btu/(sq ft)($^{\circ}$R)(hr)) compared with the effective conductivity terms $K_{eff,i}/\delta_i$ so that U_b is approximately

$$U_b \simeq \frac{1}{\displaystyle\sum_{i=1}^{N} \frac{\delta_i}{K_{eff,i}}}$$

Assume further that $K_{eff,i}$ is essentially constant and that the thickness of each layer δ_i is the same, so that δ_i is about the same for each layer. Therefore U_b becomes approximately

$$U_b \simeq \frac{K_{eff}}{\delta N} \tag{3}$$

Reference 4 suggests the following correlations for K_{eff}:

For horizontal layers:

$$\left(\frac{K_{eff}}{K}\right)_{horiz} = 0.068\left(\frac{Pr}{0.72}\, Gr_\delta\right)^{1/3}$$

where

$$Gr_\delta > 4\times10^5 \tag{4}$$

For vertical layers:

$$\left(\frac{K_{eff}}{K}\right)_{vert} = 0.065\left(\frac{Pr}{0.72}\, Gr_\delta\right)^{1/3}\left(\frac{L}{\delta}\right)^{-1/9}$$

where

$$1.1\times10^7 > Gr_\delta > 2\times10^5 \qquad \text{for} \quad 40 > \frac{L}{\delta} > 5 \tag{5}$$

The Grashof number Gr_δ is based on the layer thickness δ. These correlations were obtained for closed volume air layers at steady state. The dimensionless parameter $PrGr_\delta$ (multiple of the Prandtl and Grashof numbers) is given by

$$(PrGr_\delta) = (\chi\ \Delta T\ \delta^3) \tag{6}$$

where

$$\chi = \frac{\rho g_o c_p \beta}{\nu K} \tag{7}$$

Substituting equation (6) into equations (4) and (5) and substituting those results into equation (3) result in

$$(U_b)_{horiz} \simeq \frac{0.075\ K}{N}\,(\chi\ \Delta T)^{1/3} \tag{8}$$

for horizontal layers and

$$(U_b)_{vert} \sim \frac{0.072\ K}{N} (\chi\ \Delta T)^{1/3}\left(\frac{\delta}{L}\right)^{1/9} \tag{9}$$

for vertical layers. The empirical equations (8) and (9) indicate that the effect of tank size L and the layer thickness δ on the value of U_b is negligible for horizontal layers and small for vertical layers when $L \gg \delta$. The equations also point out that it is desirable to use as many layers as practical (N large) to decrease the heat leak.

Solutions are generated for U_b from equations (8) and (9) for the case of single-layer thermal barriers (horizontal and vertical) that contain liquid hydrogen at steady state, where $\Delta T = T_{s(p)} - T_{(t)}$ is taken at $\Delta T_{max} = T_{s(p)} - T_o$. This results in values for U_b that would be somewhat larger than those for the actual case, where $T_{(t)}$ increases from T_o as the shielded liquid is heated (see eqs. (8) and (9)). These maximum values for U_b are plotted in figure 5 as a function of tank pressure since $\Delta T_{max} = T_{s(p)} - T_o$ is only a function of pressure.

Reference 4 indicates that oblique layers may be scaled from the equations describing horizontal and vertical layers (eqs. (8) and (9)) by a linear interpolation. Therefore, the effective thermal conductance of layers at an angle θ from the horizontal $(U_b)_\theta$ can be written as

$$(U_b)_\theta \simeq (U_b)_{vert}\left(\frac{\theta}{90}\right) + (U_b)_{horiz}\left(1 - \frac{\theta}{90}\right) \tag{10}$$

Substituting equations (8) and (9) into equation (10) results in

$$(U_b)_\theta \simeq \frac{K}{N}\left[0.075\left(1 - \frac{\theta}{90}\right)\left(\chi\ \Delta T\right)^{1/3} + 0.072\left(\frac{\theta}{90}\right)\left(\chi\ \Delta T\right)^{1/3}\left(\frac{\delta}{L}\right)^{1/9}\right] \tag{11}$$

Consider now an example that will be compared with experimental data in the RESULTS AND DISCUSSION section. A thermal barrier (composed of two, $N = 2$, δ = 1-inch-thick layers, L = 1.5-foot-long liquid-hydrogen layers at $\theta = 60^\circ$ to the horizontal) will be subjected to a temperature difference $\Delta T_b = T_{s(p)} - T_{(t)}$. The thermal conductance of this barrier U_b can be determined as a function of ΔT_b and pressure from equation (11). The result of this computation is plotted in figure 6 for tank pressures of 8 and 40 pounds per square inch gage where $T_{bk(t)} = T_o$ at $t = 0$. Note that the effective thermal conductance for this arrangement U_b runs in the neighborhood of $6 < U_b < 23$ Btu per square foot per $^\circ$R per hour.

Of the assumptions used in the derivation of the equation for U_b (eqs. (8) and (9)), two assumptions stand out as sufficiently questionable to warrant experimental verification.

(1) The data used in the correlation were obtained for volumes of air at $2\times10^5 < Gr_\delta < 1.1\times10^7$ and $5 < L/\delta_i < 40$. The case considered is for nearly saturated liquid hydrogen and for Gr_δ outside this range.

(2) Convection within the liquid layers must be minimized. Consequently, no boiling can be allowed there since this would cause considerable mixing, and the effectiveness of the thermal barrier would decrease greatly. In addition, the required pressure-equalizing openings could also cause mixing.

Effectiveness of barrier: An accurate detailed evaluation of the total weight saving made possible by the barrier over the methods currently employed to reduce potential propellant losses through pump cavitation would require a detailed design study of the propellant systems. Such a study is beyond the scope of this report. An estimate of the effectiveness of the barrier can be made, however, by comparing the amount of liquid that cannot be pumped, because of liquid heating, for a tank equipped with a thermal barrier to that for the same tank under the same conditions but without a thermal barrier, as shown in figure 7. Consider the following case where a tank is suddenly pressurized, and heat leak through the tank walls or barrier heats the liquid at the wall above the initial temperature T_o. This less dense liquid flows by convection currents to the liquid-vapor interface producing the typical temperature profiles in figure 7, if no mechanical mixing of liquid occurs. For purposes of estimation, assume that all the heat to the liquid enclosed by the barrier (fig. 7(b)) or within the tank (fig. 7(a)) goes to form a layer of saturated liquid at a temperature $T_{s(p)}$, as shown by the assumed temperature profiles, and that the mass of liquid in this layer would be unusable because of detrimental pump cavitation.

The thickness of this saturated layer grows with time. The relation between the heat input to the tank and the mass growth rate of the saturated layer for a tank without a barrier is given by

$$\frac{Q}{A} A \simeq \frac{d}{dt}\int_V \rho_l h \, dV \simeq \dot{m}_l\left[h_{s(p)} - h_o\right] \tag{12}$$

Equation (12) is based on the assumption that the tank pressure is constant with time (ref. 5) and the thickness of the saturated layer is small compared with the liquid depth. If the thickness of the layer is initially zero, the mass in the layer as a function of time is given by

$$m_{l(t)} = \dot{m}_l t \approx \frac{\left(\frac{Q}{A}\right)At}{\left[h_{s(p)} - h_o\right]} \tag{13}$$

When the thermal barrier is mounted in the tank (fig. 7(b)), the heat transferred to the shielded liquid, based on equation (1), is given by

$$\left(\frac{Q}{A}\right)_b A_b \simeq U_b A_b\left[T_{s(p)} - T_o\right] \tag{14}$$

In this case, the mass converted to the warm layer is similarly derived to be

$$m_{l,b} = \dot{m}_{l,b}(t) = \frac{U_b A_b \left[T_{s(p)} - T_o\right] t}{\left[h_{s(p)} - h_o\right]} \tag{15}$$

In addition to the liquid lost by pump cavitation, some liquid is lost by sacrificial boiloff in the outer volume. The liquid boiled off when a thermal barrier is used is given by

$$\frac{t}{h_{fg}} \left\{ \frac{Q}{A} A - U_b A_b \left[T_{s(p)} - T_o\right] \right\} \simeq \dot{m}_v t \tag{16}$$

where $(Q/A)A$ is the same total heat leak input to the tank as for the case with no barrier. Although the liquid in the outer volume is warm (saturated), its volume is small, and it will mix uniformly with the cool shielded liquid during outflow. Thus, only a small amount of it, l_b thick, will be lost by cavitation. The ratio of the mass lost with a thermal barrier to that lost in a tank not equipped with a barrier Γ can be determined from the following relation:

$$\frac{\text{Mass lost with a barrier as a result of cavitation and sacrificial boiloff}}{\text{Mass lost without a barrier as a result of pump cavitation}} \equiv \Gamma$$

$$\equiv \frac{\dot{m}_{l,b} + \dot{m}_v}{\dot{m}_l} \tag{17}$$

Combining equations (13), (15), and (16) and substituting into equation (17) result in

$$\Gamma \simeq \frac{U_b A_b}{\frac{Q}{A} A} \left[T_{s(p)} - T_o\right] \left\{1 - \frac{\left[h_{s(p)} - h_o\right]}{h_{fg}}\right\} + \frac{\left[h_{s(p)} - h_o\right]}{h_{fg}} \tag{18}$$

Equation (18), plotted in figure 8, indicates that an appreciable weight saving may be possible for tanks designed for low tank pressures and high heat flux Q/A. For example, consider a large cylindrical hydrogen propellant tank where the barrier, composed of two narrow layers, $L/\delta = 100$, is mounted close to the tank wall so that $A_b \simeq A$. According to figure 5 a reasonable representative value for U_b, at a tank pressure of 10 pounds per square inch gage and a temperature T_o of 36.7° R, would be about $U_b \simeq 10$ Btu per square foot per $^\circ$R per hour. For a range of heat flux, for booster vehicles, from the environment of $50 < Q/A < 1000$ Btu per square foot per hour, the resulting range of Γ would be $0.08 < \Gamma < 0.7$.

The effectiveness of the liquid thermal barrier is further demonstrated by case D in the illustrative example of appendix B. For case D, the thermal barrier made of the foam type material of case C is replaced by a liquid thermal barrier (fig. 4) with a conductance $U_b = 10$ Btu per square foot per $^\circ$R per hour. The usable propellant fraction η for case D is plotted in figure 2. A comparison of the usable propellant fractions for these cases with other cases indicates that the liquid thermal barrier is very nearly as effective in pro-

ducing a weight saving over the standard tank design of case A as the foam type barrier used for case C and that both types of thermal barrier give usable propellant fractions very close to the ideal, as given by case B. This relative advantage of the barrier (case D) is maintained even if many of the parameters (insulation density L/D allowable tank stress and time) are varied over a practical range. Furthermore, the liquid barrier is probably more practical than the rigid barrier of case C. These savings, which are greatest at tank pressures below 20 pounds per square inch, would come from savings in liquid lost by cavitation and/or tank weight resulting from high tank pressure or increased weight of insulation to prevent this cavitation, etc.

APPARATUS AND PROCEDURE

Some questionable assumptions that warrant experimental verification were pointed out in the analysis:

(1) The analysis extrapolated experimental data for U_b to such a degree that it may not even be adequate for estimates of thermal performance in this case.

(2) No appreciable mixing in the thermal-barrier layers was allowed, which means that there could be no boiling in the barrier layers and that the openings, used to equalize the liquid level in each layer, cause no appreciable mixing there.

These assumptions were evaluated by performing tests on a simple model of a thermal barrier that could be easily fabricated and tested, and still permit the preceding assumptions and the barrier concept to be verified. The thermal-barrier model, test facility, instrumentation, and test procedure are outlined in the following section.

Apparatus

Three conical bags were fabricated of Mylar (2 mil) and attached to hoops so that the spacing between the conical bags was 1 inch (fig. 9). Four holes (5/16-in. diam) were made in each of the bags near the apex of the cone that served to equalize the liquid level within each volume. The hoops were attached to the lid of an available cryostat by long rods. A temperature rake of 33 carbon resistor thermometers was placed within the bulk liquid volume on the centerline of the cones, and two individual carbon resistor thermometers were placed in the outer volume and in each of the two thermal barrier volumes. A 1-kilowatt electric heater was provided in the outer volume to heat that liquid to boiling quickly. Windows in the lid of the cryostat allowed continuous viewing of the liquid condition in each volume during the test. The liquid level was visually monitored by noting the interface location on a ruler attached to the innermost cone.

Procedure

The tank was filled so that the liquid, when heated, would not grow out of the confines of the cones (e.g., $h_{max} \simeq 15$ in., fig. 9(a)). The tank, initially

vented to the atmosphere, was quickly pressurized and held at a constant pressure (8 or 40 psig). The electric heater was concurrently turned on to full power (i.e., zero time). When boiling in the outer layer was obtained, the heater power was reduced to a minimum to keep the liquid level near a depth h of about ≃13 inches while still maintaining a well-mixed boiling outer liquid volume. This procedure is permissible because the heat leak to the shielded bulk liquid does not depend on the heat input to the outer volume once the outer volume is saturated.

At about 5-minute intervals, from and including zero time, temperature data and tank pressure were digitally recorded. The liquid level was observed at these times. Evidence of boiling in all layers was visually monitored.

For this test, the volume of the thermal barrier and the outer volumes was necessarily an appreciable part of the total liquid volume, consequently the thermal transient behavior of these layers must be considered when the resulting experimental data are evaluated.

Instrumentation Accuracy

The temperature rake and three pairs of carbon resistor thermometers were covered by saturated boiling hydrogen ($p = 14.7$ psia) at the start of the experiment to check their thermal accuracy at one known condition. Based on this, calibration accuracy, and an estimate of the thermal conduction errors involved, thermal errors of no more than $\pm 0.1^{\circ}$ R probably resulted. The tank pressure was sensed within ±0.1 percent of full scale. Visual determination of the liquid level was within ±0.25 inch.

RESULTS AND DISCUSSION

Experimental Results

Visual observation indicated that at no time during the experiment was there any boiling in any of the thermal barrier layers or the shielded volume. This result indicates that the liquid within the thermal barrier could be sufficiently stagnant to retard heating of the shielded liquid significantly. The outer volume was quickly brought to a boil in 2.25 and 6.75 minutes for tank pressures of 8 and 40 pounds per square inch gage, respectively. If there were no barrier, all the liquid in the tank would have been boiling in about 2.6 and 7.8 minutes, respectively, whereas the barrier shielded the liquid within its confines so that saturation was not approached for more than 45 minutes.

The temperature profiles taken along the axis of the cone in the shielded liquid volume and at various times after the start of pressurization are plotted in figure 10. As expected, the liquid is temperature stratified in the shielded volume. The liquid depth increased at first as a result of liquid heating then dropped because of outer volume boiloff.

The overall thermal conductance U_b of the two-layer thermal barrier used in this experiment can be determined approximately from experimental data. In

practical situations, the volume of liquid in the thermal-barrier layers is small so that its thermal-storage capability can usually be ignored. For such a system, the thermal conductance can be determined by equation (1)

$$U_b = \frac{\left(\frac{Q}{A}\right)_b}{\Delta T_b}$$

where ΔT_b is the temperature across the barrier. A complication arises in this determination because the liquid enclosed by the barrier is not at a spatially uniform temperature, but rather it is temperature stratified (figs. 10(a) and (b)). However, these figures indicate that most of the liquid (bulk liquid) enclosed by the barrier does heat up uniformly. Since the liquid that is heated at the barrier wall comes from this bulk liquid, it appears reasonable to take ΔT_b as

$$\Delta T_b \simeq T_{s'(p)} - T_{bk(t)} \tag{19}$$

Thus equation (1) becomes

$$U_b \simeq \frac{Q_b}{\overline{A}_b\left[T_{s'(p)} - T_{bk(t)}\right]} \tag{20}$$

where A_b is the average wetted area of the two layers of the thermal barrier and Q_b is the heat transferred through the barrier with no thermal storage in the barrier layers.

An energy balance on the shielded liquid volume results in equation (21), which is based on the assumptions of negligible kinetic energy, potential energy, thermal radiation, mass transfer, and work

$$Q_b' \simeq \frac{d}{dt}\int_V \rho_l h \, dV \simeq \frac{d}{dt}\left[\sum_{i=1}^{M}\left(\rho_l c_{p_l} VT\right)_i\right] \tag{21}$$

where Q_b' is the heat transferred to the shielded liquid. Substitution of equation (21) into equation (20) can be accomplished if $Q_b' = Q_b$, which would occur in most practical situations where the thermal-barrier liquid and outer-volume liquid would have relatively negligible volume. The substitution results in

$$U_b \simeq \frac{Q_b'}{\overline{A}_b\left[T_{s'(p)} - T_{bk(t)}\right]} \simeq \frac{\frac{d}{dt}\left[\sum_{i=1}^{M}\left(\rho_l c_{p_l} VT\right)_i\right]}{\overline{A}_b\left[T_{s'(p)} - T_{bk(t)}\right]} \tag{22}$$

Unfortunately the thermal barrier studied in this experiment did not possess a relatively negligible volume so that its thermal storage should be taken into account. The best way to handle this situation is to write an approximate heat balance on the inner barrier layer (layer 2, see fig. 9).

$$U_b \simeq \frac{K_{eff}}{2\delta} \simeq \frac{1}{2\overline{A}_b(\overline{T}_1 - \overline{T}_2)} \left[\delta \frac{d}{dt}\left(\rho_l c_{p_l} \overline{AT}\right)_2 + Q_b'\right] \tag{23}$$

Substituting for Q_b' from equation (21) gives

$$U_b \simeq \frac{1}{2\overline{A}_b(\overline{T}_1 - \overline{T}_2)} \left[\delta \frac{d}{dt}\left(\overline{A}\rho_l c_{p_l} \overline{T}\right)_2 + \frac{d}{dt}\left(\sum_i^M \rho_l c_{p_l} VT\right)_i\right] \tag{24}$$

Liquid temperatures in the thermal-barrier layers were obtained for one location so that $\overline{T}_1$ and $\overline{T}_2$ are approximated by T_1 and T_2, respectively. The temperature difference $T_1 - T_2$, and thermal decay in layer 2, $d/dt(\rho_l c_{p_l} T)_2$, are plotted in figure 11. Processing the temperature, pressure, and liquid-level data, according to equation (24), result in the values of U_b plotted in figure 6. Comparison of the experimental values of U_b with those derived from equation (12), for this case, indicates that the experimental values are within 50 percent of the analytical values for U_b. The correlation used to determine U_b analytically (eq. (12)) was greatly extrapolated from data for enclosed volumes of air. For purposes of estimation, this correlation appears to be adequate for liquid hydrogen.

The Mylar cones and their support hoops exhibited no structural problems during the two test runs in liquid hydrogen. During filling, the cones tended to collapse inwardly because there was a higher liquid level in the outer volume, which was caused by the pressure-equalizing openings connecting the volumes in series. The cones straightened out when filling was terminated. This small difficulty could have been prevented by filling the cones from the inside rather than relying entirely on flow through the openings.

Practical Applications

The experiment, previously described in the section APPARATUS AND PROCEDURE, used an impractical liquid-thermal-barrier design. A more practical liquid-thermal barrier for a booster vehicle tank is proposed and briefly discussed in this section. The proposed arrangement is outlined in figure 12. The thermal barrier consists of three thin (approx. 1/2 mil) Mylar bags separated by dimples, which are heat formed in the Mylar sheets. The outermost bag is connected close to the wall of the tank by strings under tension. The strings are under tension because of the slightly higher pressure in the outer volume, which is caused by the restriction of the sacrificial boiloff gas from the outer liquid volume. The bags have openings, top and bottom, to minimize steady loads caused by steady pressure differences. They are flexible and flexibly supported so that transient loads due to pressurization, vehicle accel-

eration, sloshing, filling, etc., cause them to move to the tank wall to prevent damage to the bags (fig. 12).

From a structural-fabrication standpoint, the proposed design should be within the state of the art for the following reasons:

(1) No significant flexing of the barrier bags required.

(2) Some leakage is permissible.

(3) Bag spacing is not very critical for good thermal performance.

(4) Mylar is strong, with an ultimate strength of 20×10^3 pounds per square inch.

The weight penalty of three 1/2-mil-thick Mylar bags required as a thermal barrier in the cylindrical tank of case D is 25 pounds. Clearly, the weight penalty is small. The added weight of the support string and fasteners (fig. 12) should not change this conclusion.

The weight of pressurization systems, required to maintain a given tank pressure during outflow, can be large. This weight penalty is increased by heat transfer from the warm gas to the cold tank walls, exposed during outflow, which cools the gas so that more gas must be supplied to maintain a given pressure (fig. 13, refs. 6 and 7). The weight of a tank pressurization system could, in principle, be reduced by a thermal barrier because

(1) With the barrier, it is possible to operate at a lower pressure, without cavitation, thereby requiring less pressurizing gas.

(2) In high-heat-leak tanks the sacrificial boiloff gas could be used to decrease the amount of pressurant gas required.

(3) Warm gas, used as the pressurant, may not be able to flow to or transfer heat to the cold walls as they are uncovered by the retreating liquid during outflow, because the bags act as a thermal and hydrodynamic barrier (fig. 12). Consequently, the warm gas would not be cooled and become more dense. Therefore less pressurant may be required than for a tank not containing a barrier.

(4) Interfacial mass transfer (condensation), which increases pressurant requirements, decreases with decreasing pressure. Since the barrier would allow operation at lower pressure it appears that pressurant requirements would be less.

CONCLUDING REMARKS

An effective stagnant fluid thermal barrier that uses liquid hydrogen appears to be possible. Existing correlations for closed volumes of air can apparently be used to estimate the thermal conductance of a thermal barrier composed of layers of liquid hydrogen, when the temperature difference is not large. The heat-leak rate to the liquid shielded by a thermal barrier is inde-

pendent of heat-leak rate to the tank and can be appreciably smaller than the heat-leak rate to the tank for low tank pressures. A weight saving appears to be possible in liquid-hydrogen booster vehicle tanks, designed to use the thermal barrier, where the tank pressure can be low.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, September 28, 1965.

APPENDIX A

SYMBOLS

A	wetted area, sq ft
A_i	surface area of interface, sq ft
c_p	specific heat at constant pressure, Btu/(lb mass)($^{\circ}$R)
D	tank diameter, ft
d	thickness of insulation on wall, ft
Gr	Grashof number based on δ
g	acceleration due to gravity, 32.2 ft/sec^2
$\mathcal{H}$	surface coefficient of heat transfer, Btu/(sq ft)($^{\circ}$R)(hr)
h	depth of shielded volume liquid above tip of inner cone, ft
$\hbar$	specific enthalpy, Btu/lb mass
$\hbar_{fg}$	heat of vaporization, Btu/lb mass
K	thermal conductivity, Btu/(ft)($^{\circ}$R)(hr)
$K_{eff,i}$	effective thermal conductivity of i^{th} thermal-barrier layer, Btu/(ft)($^{\circ}$R)(hr)
L	length of thermal barrier or tank, ft
l	depth of warm layer in tank, which cannot be pumped because of cavitation, when no thermal barrier is in tank, ft
l_b	depth l when a thermal barrier is in tank, ft
M	number of thermometers in liquid
m	mass, lb mass
$\dot{m}$	mass transfer rate, lb mass/hr
N	number of fluid layers in thermal barrier
Pr	Prandtl number
p	absolute tank pressure, psia
p'	gage tank pressure, psig

Q heat-transfer rate, Btu/hr

Q/A heat leak, Btu/(sq ft)(hr)

S working stress, psi

T temperature, $^{\circ}R$

ΔT temperature difference across layer of barrier, $^{\circ}R$

ΔT_b temperature difference across barrier, $^{\circ}R$

t time, hr

U_b overall thermal conductance of thermal barrier, Btu/(sq ft)($^{\circ}R$)(hr)

V volume, cu ft

w/A weight per unit surface area, lb/sq ft

α volume fraction of liquid in tank

β compressibility factor, $(1/\rho)(\partial\rho/\partial T)_p$, $1/^{\circ}R$

Γ ratio of mass not pumped when a thermal barrier is used to that not pumped when a thermal barrier is not used

δ thickness of thermal-barrier layer, ft

η ratio of usable propellant mass to filled tank mass

θ angular inclination of barrier layers with respect to horizontal, deg

ν kinematic viscosity, sq ft/sec

ρ density, lb mass/cu ft

χ $\equiv \dfrac{PrGr_\delta}{\Delta T\delta^3} = \dfrac{\rho c_p \beta g_o}{\nu K}$, $1/(^{\circ}R)(cu\ ft)$

$\sum_{i=1}^{M}$ summation from $i = 1$ to $i = M$

Subscripts:

b bag

bk bulk liquid where temperature is spatially uniform

fg	heat of vaporization
g	gas
horiz	horizontal layer of fluid
i	summing subscript
ins	insulation
lo	lost
l	liquid
m	metal wall
max	maximum or worst case
o	initial condition
(p),(t)	functions of pressure and time, respectively
s	saturation
s'	saturation - outer volume
sl	shielded liquid
t	tank
v	boiloff vapor
vert	vertical layer of fluid
w	wetted
∞	outside environment of tank
1,2	outer and inner barrier layers, respectively

Superscript:

$(\bar{\ })$	spatially averaged value

APPENDIX B

CALCULATIONS FOR COMPARISON OF CAVITATION PREVENTION METHODS

A highly simplified analysis of the effectiveness of various cavitation prevention methods is presented. This estimate is made for a given tank with the insulation thickness and tank pressure varied. Four general cases will be considered (see fig. 2 for schematic drawings):

Cases A: Simply insulated tank for varying insulation thickness

Case B: Ideal tank where heat is completely converted to boiloff vapor without affecting the bulk liquid

Case C: Insulated tank with a thermal barrier composed of foam (fig. 3)

Case D: Insulated tank with a thermal barrier composed of liquid-filled bags (fig. 4)

The following simplifying assumptions were made for all cases:

(1) The heat leak is uniform over the wetted area. Interfacial heat and mass transfer, which increase with increased pressure, are neglected.

(2) The heated liquid flows to form a relatively shallow saturated layer of liquid below the interface. This layer is considered unusable because of pump cavitation.

(3) The tank is taken to be a flat-ended cylindrical metal tank whose wall thickness is uniform and whose thickness is determined by the hoop stress

$$d_m = \frac{(p - p_\infty)D}{2S} \tag{B1}$$

The insulation is also of uniform thickness.

Cases A

The heat input to the tank is

$$\frac{Q}{A} A_w \cong U_b(T_\infty - T_o)\left(\frac{\pi D^2}{4} + \alpha\pi DL\right) \tag{B2}$$

where

$$U_b \simeq \frac{1}{\frac{1}{\mathcal{H}_\infty} + \frac{d}{K_{ins}}} \tag{B3}$$

therefore

$$\frac{Q}{A} A_w \cong \frac{(T_\infty - T_o)\frac{\pi D^2}{4}}{\frac{1}{\mathcal{H}_\infty} + \frac{d}{K_{ins}}}\left(1 + 4\alpha \frac{L}{D}\right) \tag{B4}$$

This heat, by assumption (2), raises the total enthalpy of the saturated liquid layer through increased thickness, l:

$$\frac{Q}{A} A_w = \rho_{l_s} \frac{\pi D^2}{4}\left(h_{s(p)} - h_o\right)\frac{dl}{dt} \tag{B5}$$

Combining equations (B4) and (B5) and integrating ($l = 0$ at $t = 0$) result in the mass of liquid lost by cavitation:

$$m_{lo} \equiv \rho_{l_s} \frac{\pi D^2}{4} l(t) = \frac{\pi D^2}{4} \frac{(T_\infty - T_o)\left(1 + 4\alpha \frac{L}{D}\right)t}{\left(\frac{1}{\mathcal{H}_\infty} + \frac{d}{K_{ins}}\right)\left(h_{s(p)} - h_o\right)} \tag{B6}$$

Based on the hoop stress and $p_\infty = 0$, the mass of a tank of uniform wall thickness is

$$m_t = \rho_m \frac{\pi D^2}{4}\left(2 + 4\frac{L}{D}\right)\frac{pD}{2S} \tag{B7}$$

The mass of the uniform insulation covering is

$$m_{ins} = \rho_{ins} d \frac{\pi D^2}{4}\left(2 + 4\frac{L}{D}\right) \tag{B8}$$

The ratio of usable propellant mass to the filled tank mass η is defined as

$$\eta \equiv \frac{m_l - m_{lo}}{m_t + m_{ins} + m_l} \tag{B9a}$$

or

$$\eta = \frac{\frac{\pi D^2}{4}\left[\alpha \rho_{l_o} L - \rho_{l_s} l\right]}{m_t + m_{ins} + \alpha \rho_{l_o} L \frac{\pi D^2}{4}} \tag{B9b}$$

Combining equations (B6), (B7), (B8), (B9a), and (B9b) results in

$$\eta = \frac{\alpha\rho_{l_o} L - \dfrac{(T_\infty - T_o)\left(1 + 4\alpha \dfrac{L}{D}\right)t}{\left(\dfrac{1}{\mathcal{H}_\infty} + \dfrac{d}{K_{ins}}\right)\left[h_{s(p)} - h_o\right]}}{\rho_m\left(2 + 4\dfrac{L}{D}\right)\dfrac{D}{2}\dfrac{p}{S} + \rho_{ins} d\left(2 + 4\dfrac{L}{D}\right) + \alpha\rho_{l_o} L} \tag{B10}$$

Case B

All the heat input for this case is converted to boiloff vapor without affecting the bulk liquid. This boiloff mass is lost. Equation (B5) of cases A is changed to

$$\dot{m}_{l_o} h_{fg(p)} = \frac{Q}{A} A_w t \tag{B11}$$

Accordingly, equation (B10) is changed to

$$\eta = \frac{\alpha\rho_{l_o} L - \dfrac{(T_\infty - T_o)\left(1 + 4\alpha \dfrac{L}{D}\right)t}{\left(\dfrac{1}{\mathcal{H}_\infty} + \dfrac{d}{K_{ins}}\right)\left[h_{fg(p)}\right]}}{\rho_m\left(2 + 4\dfrac{L}{D}\right)\dfrac{D}{2}\left(\dfrac{p}{S}\right) + \rho_{ins} d\left(2 + 4\dfrac{L}{D}\right) + \alpha\rho_{l_o} L} \tag{B12}$$

Cases C and D

Where there is a thermal barrier, the mass lost is a result of the heated liquid that would cause pump cavitation and sacrificial boiloff. The total mass of liquid lost for cases C and D can be determined by

$$m_{l_o} = \Gamma m_{l_o(\text{no barrier})} \tag{B13}$$

where Γ is defined by equation (17), evaluated by equation (18), and plotted in figure 8. To obtain values of Γ from figure 8, values of $[(Q/A)/U_b](A/A_b)$ must be determined. If the barrier is close to the tank wall, $A/A_b \simeq 1$, and if tank pressures are low, $T_\infty - T_s \simeq T_\infty - T_o$,

$$\frac{\dfrac{Q}{A}}{U_b}\frac{A}{A_b} \simeq \frac{\dfrac{Q}{A}}{U_b} \simeq \left[\frac{T_\infty - T_o}{\left(\dfrac{1}{\mathcal{H}_\infty} + \dfrac{d}{K_{ins}}\right)\dfrac{K_{ins}}{d_b}}\right]_{\text{Case C}} \simeq \left[\frac{T_\infty - T_o}{\left(\dfrac{1}{\mathcal{H}_\infty} + \dfrac{d}{K_{ins}}\right)U_b}\right]_{\text{Case D}} \tag{B14}$$

The insulation weight per unit surface area is given by

$$\left(\frac{w}{A}\right)_{ins} \simeq \rho_{ins}(d + d_b) \tag{B15}$$

for case C, the foam type insulation barrier, and

$$\left(\frac{w}{A}\right)_{ins} \simeq 3\rho_b d_b + \rho_{ins} d \tag{B16}$$

for case D, the liquid thermal barrier of three bags.

The usable propellant fraction is then

$$\eta = \frac{\alpha\rho_{l_o}L - \Gamma\left[\frac{(T_\infty - T_o)\left(1 + 4\alpha\frac{L}{D}\right)t}{\left(\frac{1}{\mathcal{H}_\infty} + \frac{d}{K_{ins}}\right)\left[h_{s(p)} - h_o\right]}\right]}{\left(\frac{w}{A}\right)_{ins}\left(2 + 4\frac{L}{D}\right) + \rho_m\left(2 + 4\frac{L}{D}\right)\frac{D}{2}\frac{p}{S} + \alpha L\rho_{l_o}} \tag{B17}$$

where Γ and $(w/A)_{ins}$ are determined by the appropriate equations for each case.

The useful propellant fraction η for each of the cases just discussed was calculated by using the following numerical values of parameters,

$$t = 2/60 \text{ hr}$$

$$S = 4.25\times10^4 \text{ psi}$$

$$\mathcal{H}_\infty = 10 \text{ Btu/(sq ft)(hr)}$$

$$p_o = 15 \text{ psia}$$

$$p_\infty = 0 \text{ psia}$$

$$T_\infty - T_o = 500^\circ \text{ R}$$

$$D = 15 \text{ ft}$$

$$L = 30 \text{ ft}$$

$$\alpha = 0.9$$

$$\rho_{l_o} = 4.4 \text{ lb/cu ft, (liquid hydrogen)}$$

$$\rho_b = 100 \text{ lb/cu ft, (Mylar)}$$

$$\rho_m = 500 \text{ lb/cu ft}$$

$$K_{ins} = 0.01 \text{ Btu/(ft)(hr)}(^{o}\text{R}), \text{ (foam)}$$

$$\rho_{ins} = 5 \text{ lb/cu ft, (foam)}$$

$$U_b = 10 \text{ Btu/(sq ft)}(^{o}\text{R})\text{(hr)}$$

Insulation thicknesses

Case A: $d = \frac{1}{8}, \frac{1}{4}, 1$ and 2 inches

B: $d = \frac{1}{8}$ or $\frac{1}{4}$ inch

C: $d = \frac{1}{8}$ inch, $\delta = \frac{1}{8}$ inch

D: $d = \frac{1}{8}$ inch, $d_b = 6\times10^{-4}$ inch (Mylar film thickness)

and the results are presented in figure 2.

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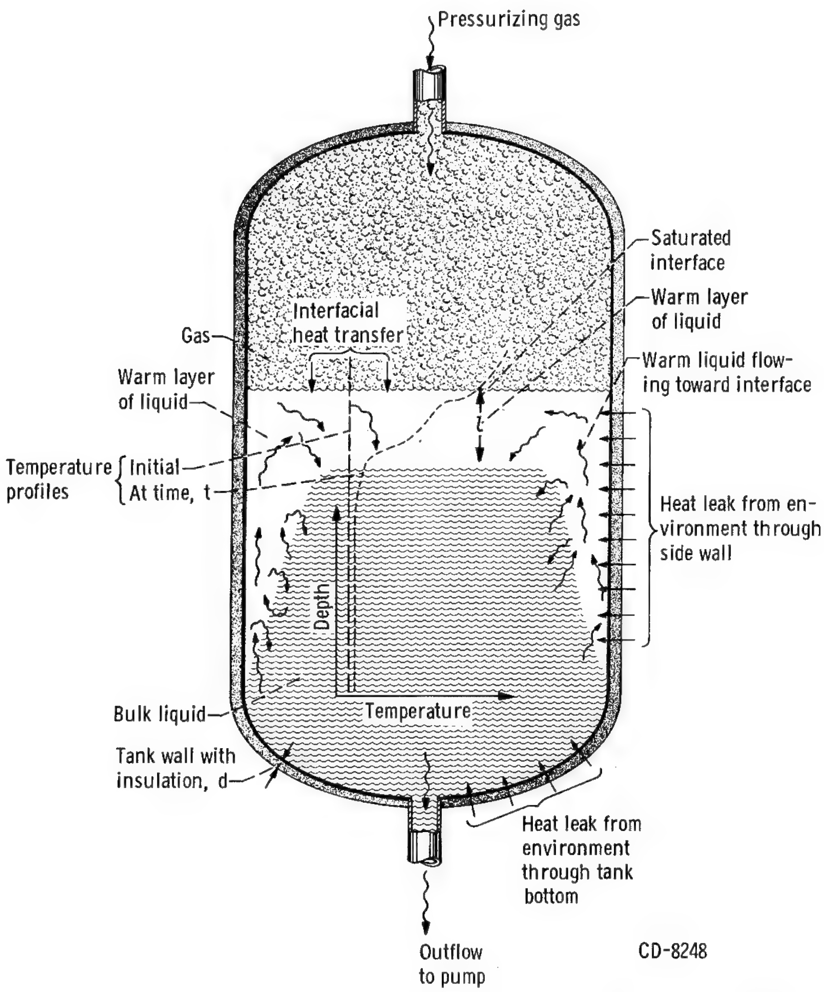


Figure 1. - Schematic drawing of liquid and heat flows during outflow from pressurized tank.

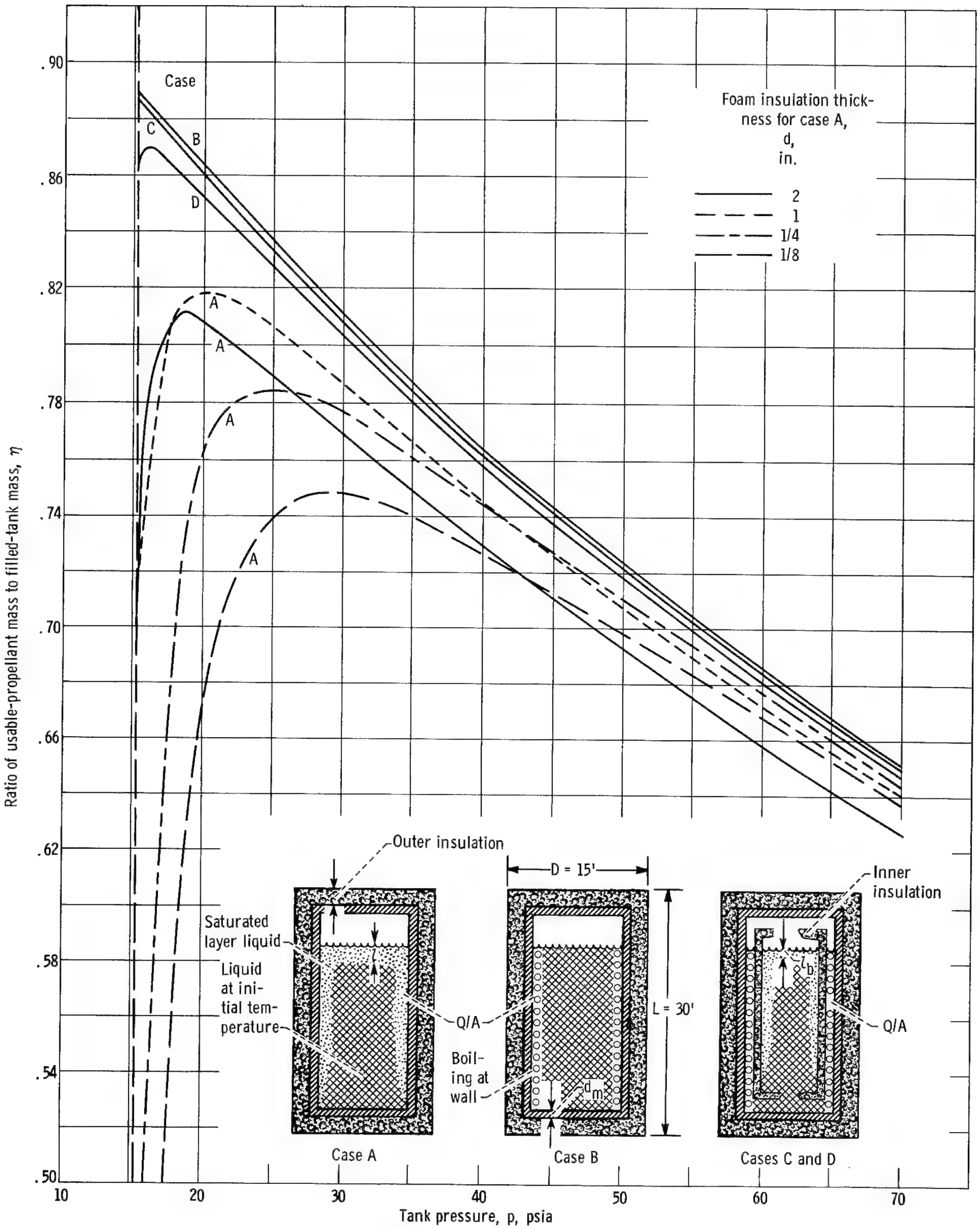


Figure 2. - Idealized comparison of cavitation prevention methods.

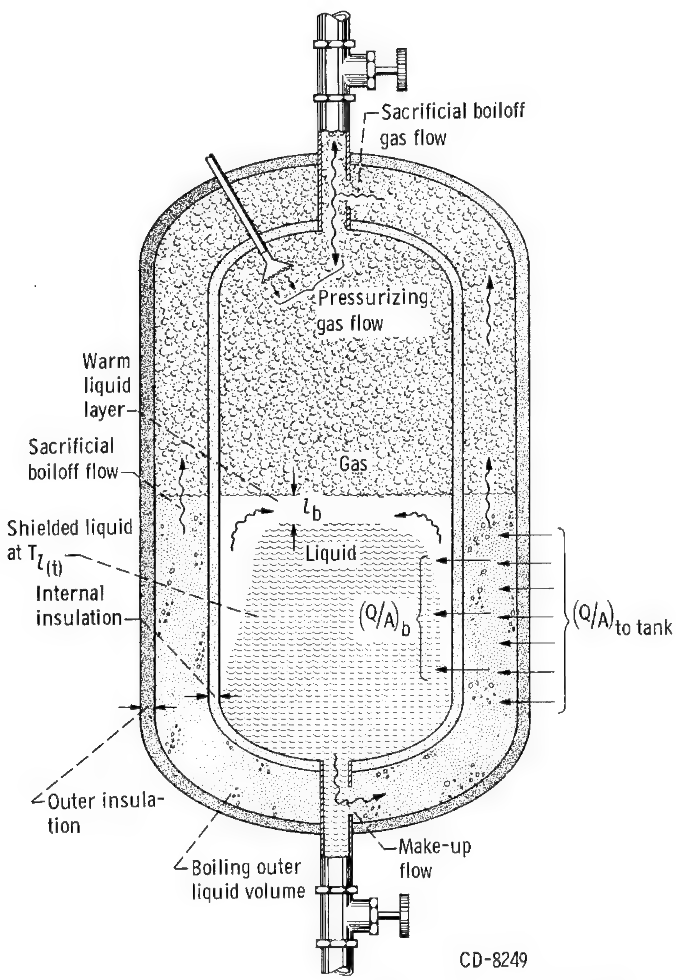


Figure 3. - Schematic drawing of method of using sacrificial boiloff to reduce rate of liquid heating.

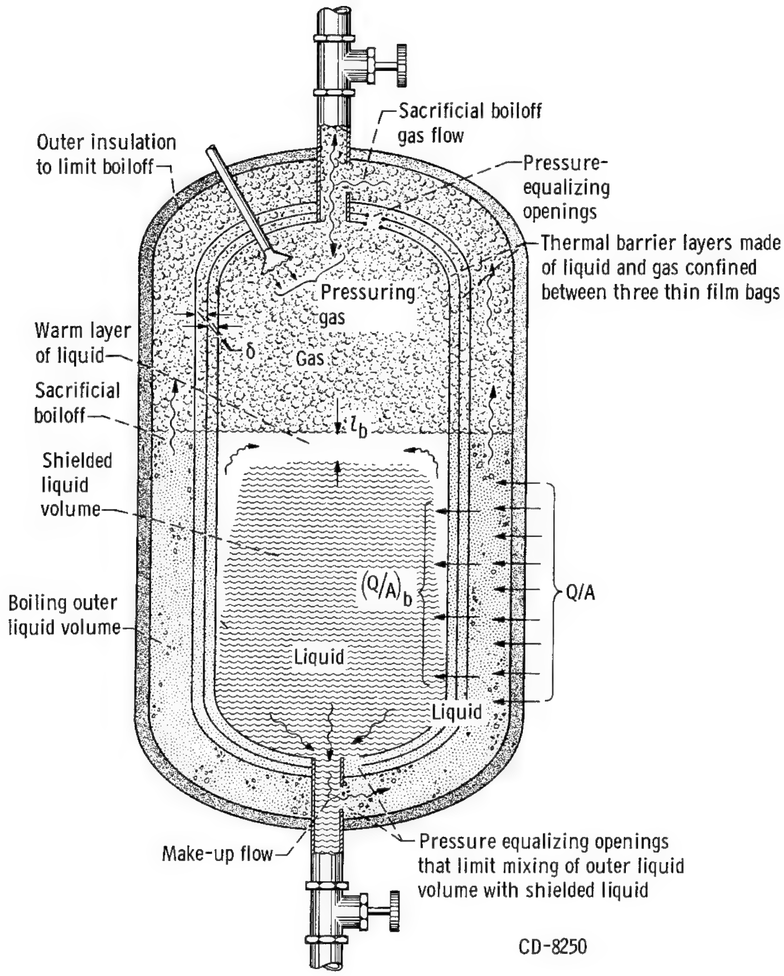


Figure 4. - Thermal barrier principle that uses liquid as insulation.

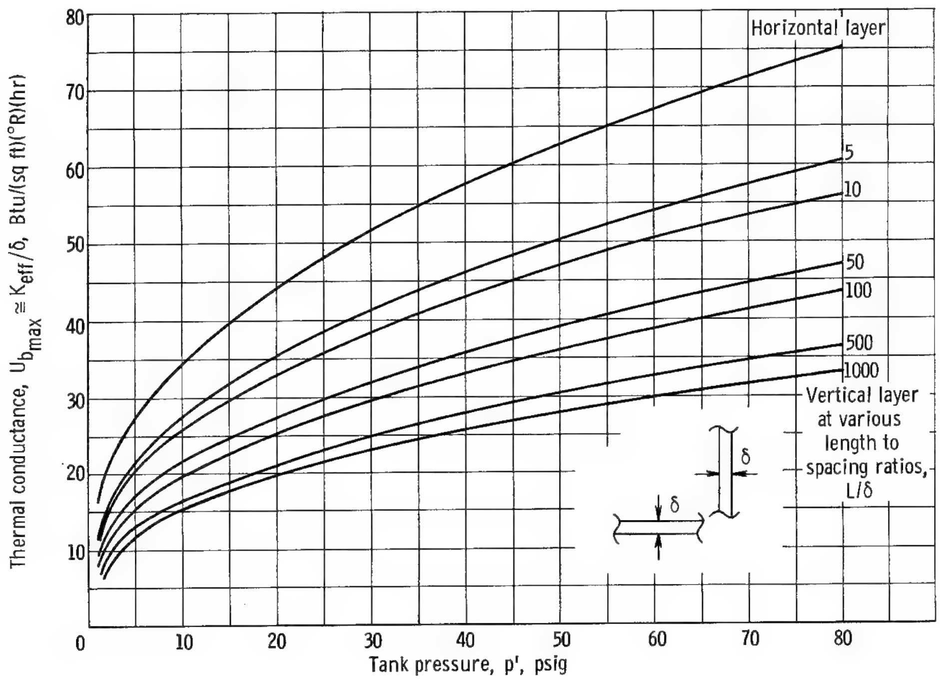


Figure 5. - Maximum thermal conductance of single layer of liquid hydrogen subjected to maximum temperature difference of $\Delta T_{max} = T_{s(p)} - T_0$.

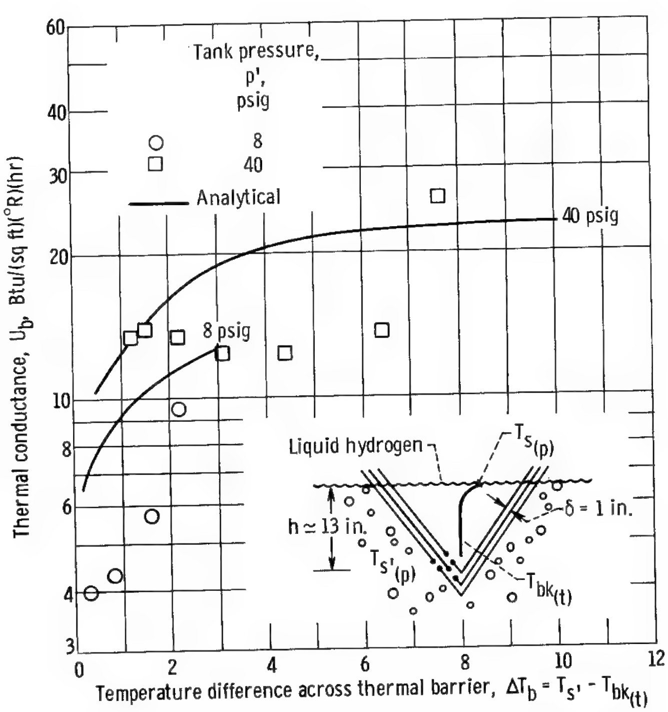
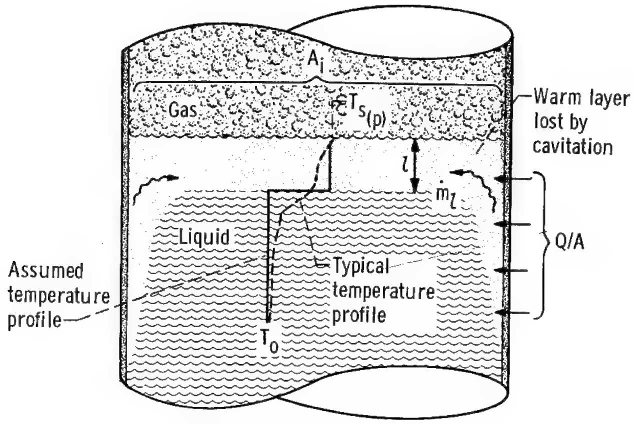
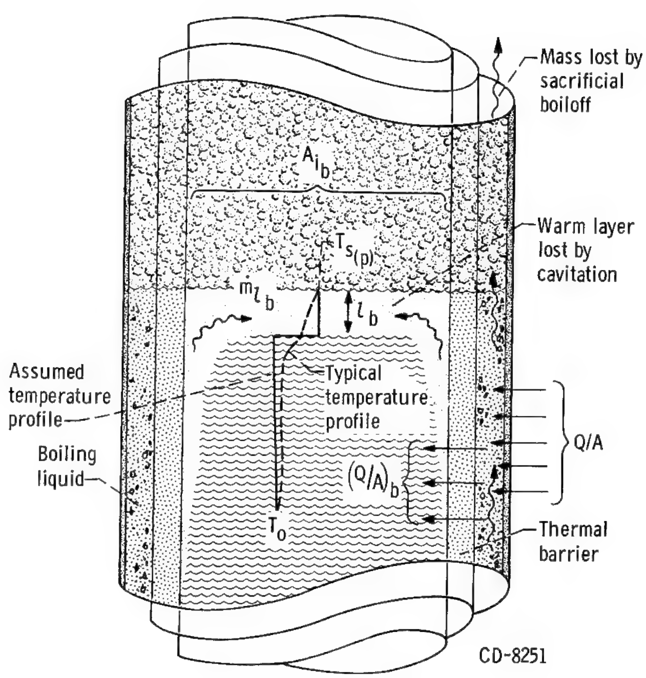


Figure 6. - Analytical and experimental values of overall thermal conductance for two-layer conical thermal barrier described in inset.



(a) Without thermal barrier.



(b) With thermal barrier.

Figure 7. - Idealized model of conditions in pressurized tanks with heat leak from environment.

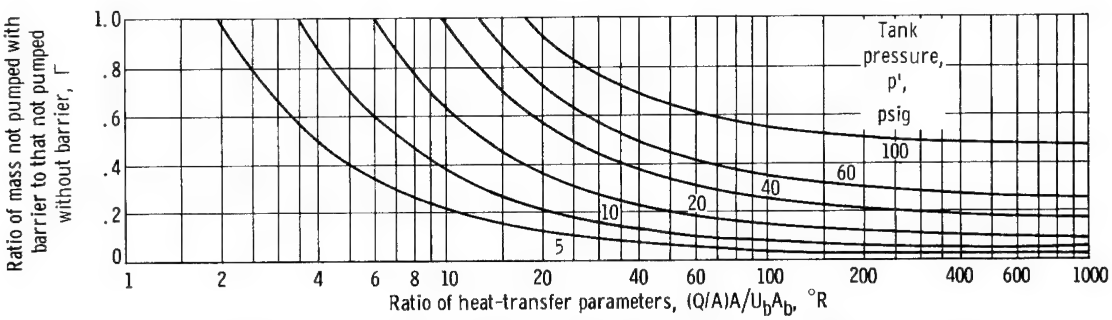
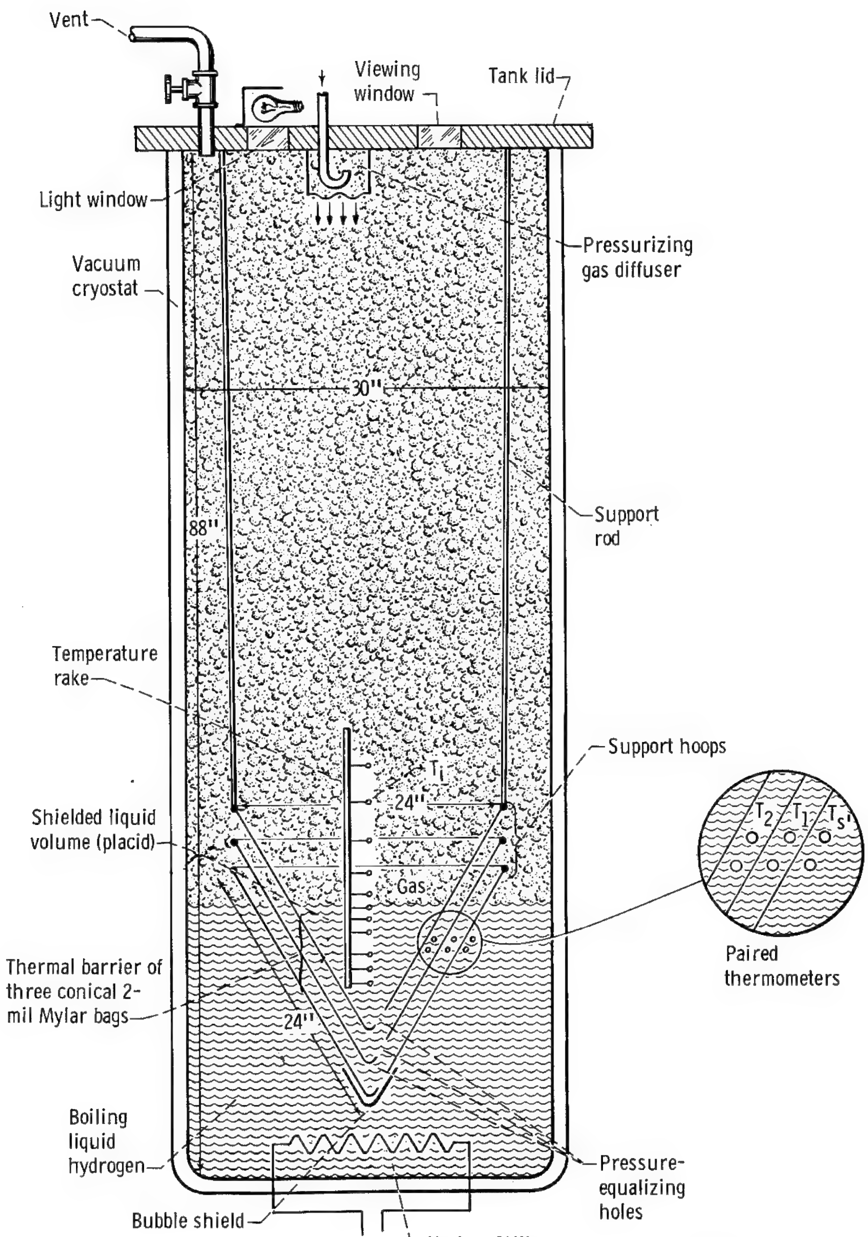


Figure 8. - Comparison of liquid losses from a tank with and without a thermal barrier (eq. 18).



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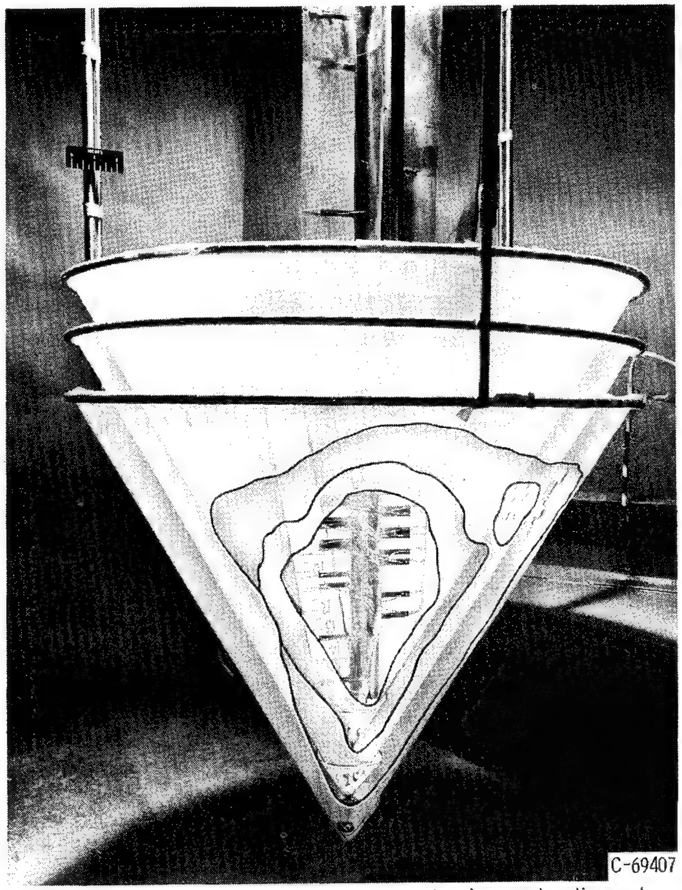
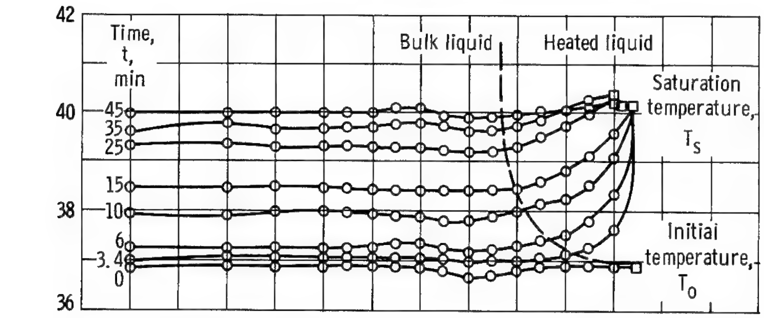
(a) Schematic drawing of test setup.

Figure 9. - Thermal barrier experimental apparatus.

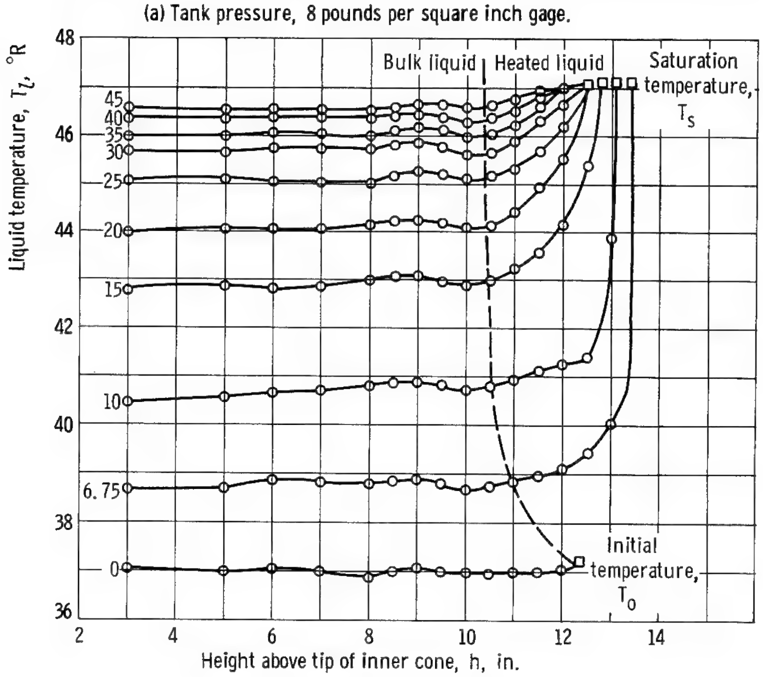


(b) Areas of bags have been cut away to show barrier construction and instrumentation.

Figure 9. - Concluded.



(a) Tank pressure, 8 pounds per square inch gage.



(b) Tank pressure, 40 pounds per square inch gage.

Figure 10. - Experimental temperature profiles at centerline of inner cone.

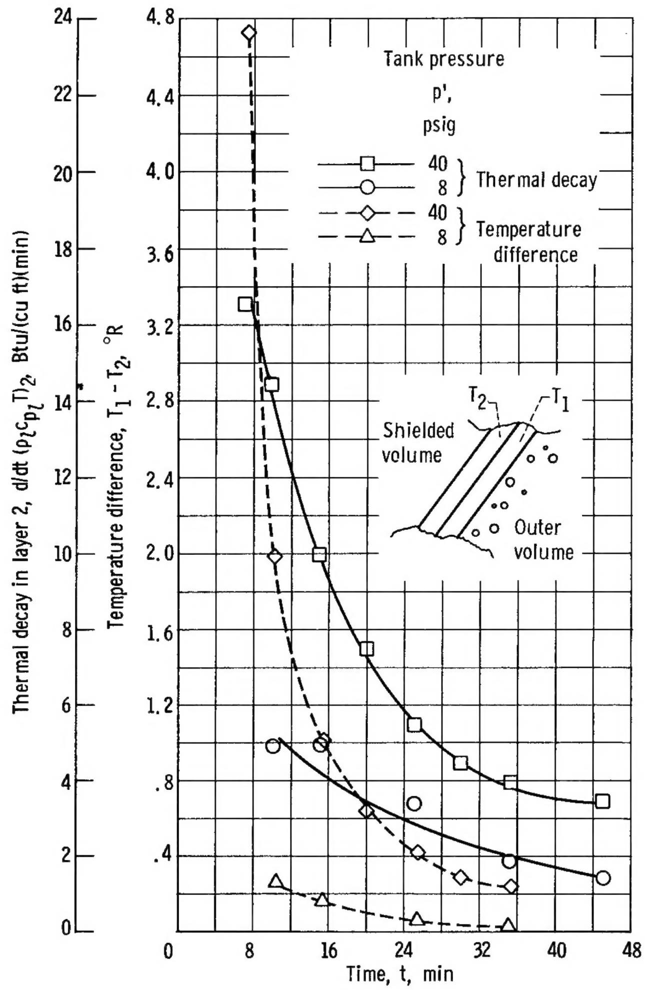


Figure 11. - Thermal data for barrier layers derived from temperature data and used in equation (24).

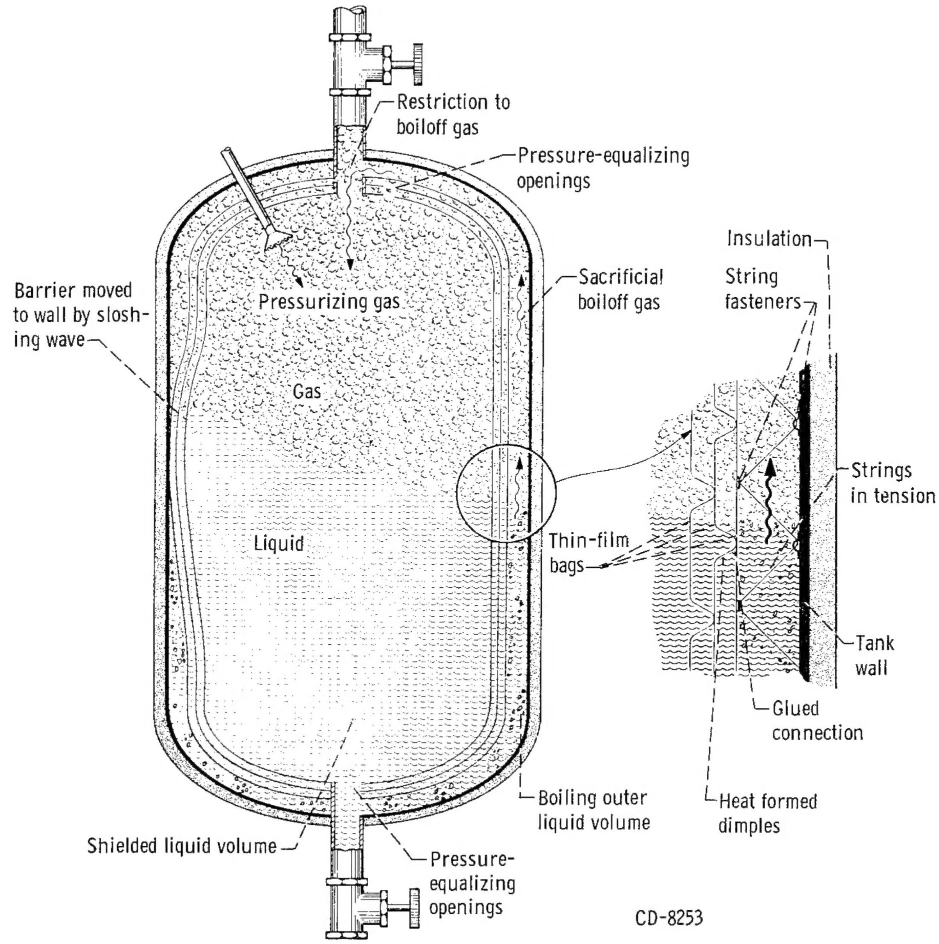


Figure 12. - Proposed thermal-barrier application in propellant tank.

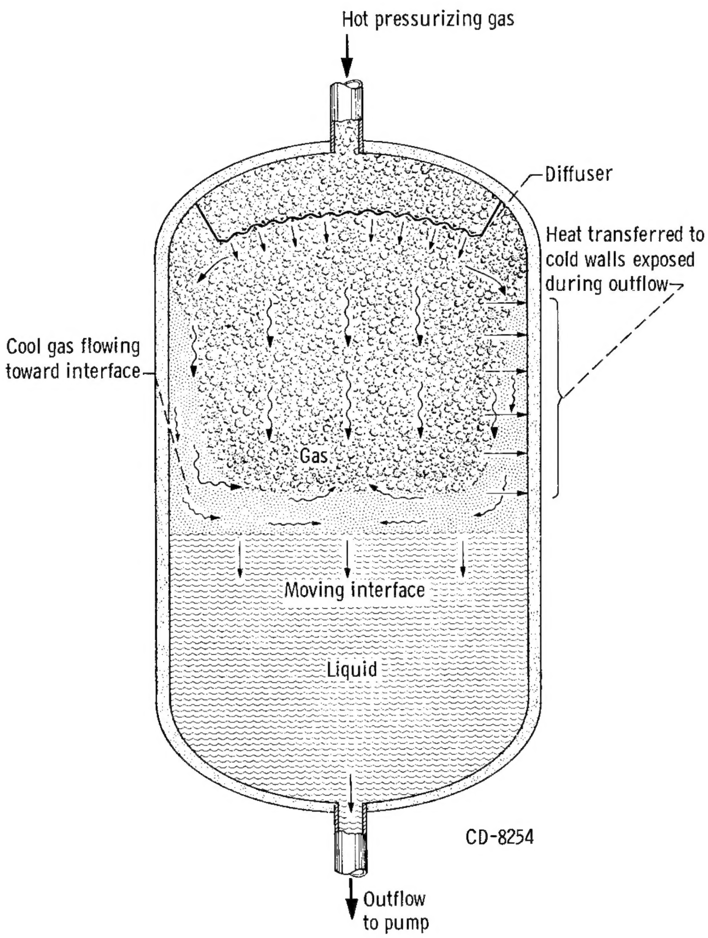


Figure 13. - Schematic drawing of gas flows during pressurized outflow of cryogenic liquid.

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—NATIONAL AERONAUTICS AND SPACE ACT OF 1958

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